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SUMMARY

This study was undertaken to determine the general geometry characteristics of alkali-metal turbines as they are affected by blade-speed limitations that might result from moisture-caused erosion effects. Selection of turbine temperature ratio was made on the basis of reduced moisture levels in the turbine. Dual turbines with intermediate moisture separation as well as single turbines were studied. Rotative speeds were so selected as to be compatible with common alternator frequencies. Turbine-geometry characteristics were determined for 1-megawatt power systems using sodium, potassium, rubidium, and cesium as working fluids at a turbine-inlet temperature of 2500° R.

The requirements for favorable turbine geometry set a lower limit on blade speed. For the conditions of this study, these lower limits were in the 250- to 400-feet-per-second range. The required number of turbine stages depends to a large extent on blade speed, as well as on the selected fluid. Reduction of blade speed results in a rapidly increasing number of stages. Blade speeds, however, can be limited to values below about 500 feet per second while geometrically feasible turbines are still maintained. Designs featuring both small and large numbers of stages were evolved. With a small number of stages, however, turbine performance can be considerably poorer than with a large number of stages. The selection of a particular design becomes the choice of the system designer, who must work within the constraints of his particular system.

INTRODUCTION

One of the indirect power-conversion systems considered for advanced space applications utilizes a Rankine (vapor-liquid) cycle with one of the alkali metals as the working fluid. In this system, mechanical shaft power is produced by the fluid expanding through a turbine. The equilibrium state point of the fluid after expansion lies in the

two-phase (vapor-liquid) region for most systems that have been studied. If the fluid reaches vapor-liquid equilibrium during the expansion process, moisture will be present in the turbine. An analytical study of the expansion-condensation behavior of the alkali metals (ref. 1) indicated that supersaturation does not persist to any great extent and moisture is readily formed, especially at the temperature level of interest (about 2500° R at the turbine inlet).

In addition to causing a performance penalty, the presence of moisture in the turbine creates a potential erosion problem, which, for any given fluid, would increase in severity with increasing blade speed. The magnitude of this problem is unknown at the present time, and imposing blade-speed limitations may be necessary in order to meet the long-time reliability requirements of space power systems. Previous studies (refs. 2 and 3) of alkali-metal turbine-geometry characteristics recognized the existence of possible blade-speed limitations resulting from moisture-caused erosion effects but imposed none on the analyses. Blade speeds, consequently, were determined solely on the basis of maximum allowable stress, and the resultant allowable blade speeds were high, often in excess of 1000 feet per second. In addition, a favorable exit hub- to tip-radius ratio was specified, and rotative speeds were allowed to vary without regard to any possible restrictions imposed by alternator frequency requirements. The turbine characteristics determined in these studies, therefore, serve to define the limit of best achievable geometries, and it must be recognized that blade-speed limitations will result in geometries less favorable than those indicated.

The study reported herein was undertaken for the purpose of determining the effect of blade-speed limitations on the general geometry characteristics of the turbine. The turbine temperature ratio was selected on the basis of reduced moisture levels, and consideration was given to dual turbines with intermediate moisture separation as well as to single turbines. Rotative speeds were so selected as to be compatible with common alternator frequencies. Turbine characteristics, such as number of stages, diameter, and exit hub- to tip-radius ratio, were determined parametrically as a function of blade speed for 1-megawatt power systems using sodium, potassium, rubidium, and cesium as working fluids at a turbine-inlet temperature of 2500° R. Several typical design points were then selected for a more refined analysis.

SYMBOLS

- A area, sq ft
- D mean-section blade diameter, ft
- E ratio of last-stage rotor- to stator-exit axial kinetic energies

g	gravitational constant, 32.2 ft/sec^2
Δh	specific work, Btu/lb
J	mechanical equivalent of heat, 778 ft-lb/Btu
m	moisture fraction
N	rotative speed, rpm
P	power, kW
r	radius, ft
U	mean-section blade speed, ft/sec
V	gas velocity, ft/sec
w	weight flow, lb/sec
α	mean-section stator-exit angle measured from axial direction, deg
η	efficiency
λ	stage speed-work parameter, $U^2/gJ \Delta h$
\bar{v}	overall blade- to jet-speed ratio, $U/\sqrt{2gJ \overline{\Delta h}_{id}}$
ρ	two-phase fluid density, lb/cu ft

Subscripts:

A	alternator
h	hub
id	ideal
n	last stage
re	rotor exit
s	static
se	stator exit
T	turbine
t	tip
u	tangential component
x	axial component

Superscript:

$-$	overall
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METHOD OF ANALYSIS

The extent of blade-speed limitations resulting from erosion considerations is unknown at this time. This analysis of turbine-geometry characteristics (number of stages, diameter, and hub- to tip-radius ratio) was therefore made with blade speed as the prime independent variable in order to cover a range of possible blade-speed limitations. The underlying philosophy of the design criteria selection is discussed before the calculation procedure is presented.

Design Criteria

The design criteria required for this analysis are cycle selection, turbine arrangement, turbine type, and rotative speed.

Cycle selection. - The vapor was specified to be saturated as it entered the turbine. As shown in reference 4, for a constant turbine-inlet temperature, superheating the vapor prior to turbine expansion is not desirable. The moisture level is reduced by only

a small amount, whereas the radiator area is increased significantly.

The severity of the potential erosion problem for any given fluid can be reduced through a reduction in turbine moisture level. As pointed out in reference 4, a reduction in moisture without a significant increase in radiator area can be achieved by increasing turbine temperature ratio above the value that yields minimum radiator area. The effect of turbine temperature ratio on both prime radiator area and turbine-exit moisture is illustrated in figure 1 for a turbine-inlet temperature of 2500°R and a turbine efficiency of 0.75. Figure 1 was obtained from equation (16) of reference 4 and Mollier charts based on the data of reference 5. The optimum turbine temperature ratio for the specified conditions lies between 0.75 and 0.80. A 0.05 increase in turbine temperature ratio yields a 0.02 to 0.03 decrease in exit moisture.

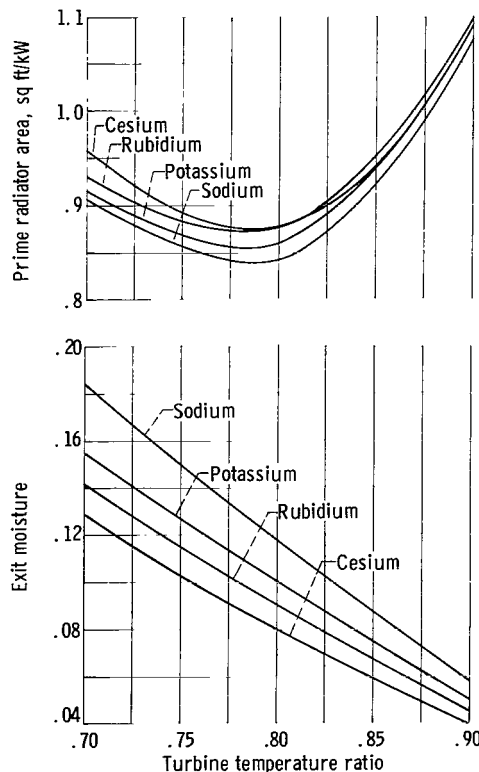


Figure 1. - Effect of turbine temperature ratio on radiator area and turbine-exit moisture. Turbine-inlet temperature, 2500°R ; turbine efficiency, 0.75.

For this study, a turbine temperature ratio of 0.80 was selected in order that some reduction in turbine-exit moisture might be gained while oper-

ation was still in the region of minimum radiator area. A turbine-inlet temperature of 2500°R was selected as representing an achievable compromise between low radiator area and turbine life (ref. 2).

Turbine arrangement. - Another way to reduce turbine-exit moisture is to use a dual-turbine arrangement, that is, two turbines with a moisture separator between them. In this way (see fig. 1) the amount of moisture in the exhaust from each turbine (turbine temperature ratio equal to about 0.90 for each turbine) would be about half of that for a single-turbine arrangement. Moisture removal in the separator was assumed to be complete, and the temperature of the vapor entering the second turbine was assumed to be equal to that leaving the first turbine. For this study, both single- and dual-turbine arrangements were considered. Interstage removal of moisture was not considered herein because experience indicates that this method is not an effective means of moisture removal (ref. 6).

Turbine type. - In order to explore a spectrum of possible turbine geometries, three types of turbines were considered: reaction, impulse, and velocity compounded. Reaction turbines, in general, are highly efficient but require a large number of stages. Velocity-compounded turbines require considerably fewer stages to do a given job but operate with a significantly lower efficiency. Impulse turbines are intermediate between the other two types both in efficiency and in number of stages. The reaction and the impulse turbines considered herein were specified to have constant stage speed-work parameters λ of 1.0 and 0.5, respectively, while the velocity-compounded turbine was specified to have a last-stage speed-work parameter of 0.5. A constant blade mean-section speed was assumed for each turbine. Additional assumptions for all turbines included a stator-exit angle of 70° and a last-stage ratio of rotor- to stator-exit axial kinetic energies of 1.5. As a result of these assumptions, the leaving loss for the impulse and the velocity-compounded turbines was greater than that for the reaction turbine. The effect of this difference in leaving loss on turbine efficiency was examined and was about four points for a single-stage turbine and less for multistage turbines. This difference does not significantly affect the results of the analysis; for simplicity, therefore, these assumptions were maintained.

Rotative speed. - If blade speeds are to be limited by erosion considerations, relatively low values of rotative speed must be selected in order to achieve reasonable turbine geometries. In addition, it is desirable to have these rotative speeds compatible with common alternator frequencies in order to be able to utilize a direct-drive system. Rotative speeds of 8000 and 12 000 rpm were chosen as relatively low speeds compatible with alternator frequencies of 400, 800, and 1000 cps with the exception that 8000 rpm is not compatible with 1000 cps. In addition, a somewhat higher rotative speed of 20 000 rpm, which is compatible with alternator frequencies of 1000 and 2000 cps, was also given consideration.

Calculation Procedure

The calculation procedure consisted of applying the specified design criteria and determining the turbine-geometry characteristics as a function of blade speed. For the dual-turbine arrangement, a turbine temperature ratio of 0.90 across the first turbine resulted, within the accuracy of reading the Mollier charts, in an equal total work ($w \Delta h$) split for the two turbines. The analysis procedure used herein is of the same general type as that of reference 2, and the equations are presented with a minimum of explanation.

Turbine weight flow is

$$w = \frac{P_A}{1.054 \eta_A \bar{\eta}_{T,s} \Delta h_{id}} \quad (\text{lb/sec}) \quad (1)$$

The alternator power output P_A is 1000 kilowatts for the single-turbine arrangement and 500 kilowatts for each turbine of the dual-turbine arrangement. The alternator efficiency η_A is assumed to be 0.85. The ideal specific work Δh_{id} is obtained from Mollier charts based on the data of reference 5. For the dual-turbine arrangement, ideal work is determined separately for each turbine rather than on the basis of a single ideal process. For simplicity in the parametric study, a constant turbine efficiency $\bar{\eta}_{T,s}$ of 0.75 is assumed for the computation of weight flow. This simplification does not yield any gross errors in the results. A better estimation for weight flow, as was made for several selected turbine designs, can be obtained by using an aerodynamic efficiency from which a moisture penalty is subtracted. In this analysis, the moisture penalty was assumed to be one point of efficiency for each point of exit moisture.

A procedure for relating overall aerodynamic static efficiency to number of stages and blade- to jet-speed ratio is presented in reference 7. The overall blade- to jet-speed ratio is defined as

$$\bar{v} = \frac{U}{\sqrt{2gJ \Delta h_{id}}} \quad (2)$$

The performance characteristics for the reaction and the impulse turbines can be obtained directly from the equations of reference 7, whereas those for the velocity-compounded turbines are obtainable through slight modifications of those equations. These performance characteristics are presented in figure 2. The previously mentioned increase in efficiency with an increase in number of stages among the selected turbine types can be evaluated quantitatively from this figure.

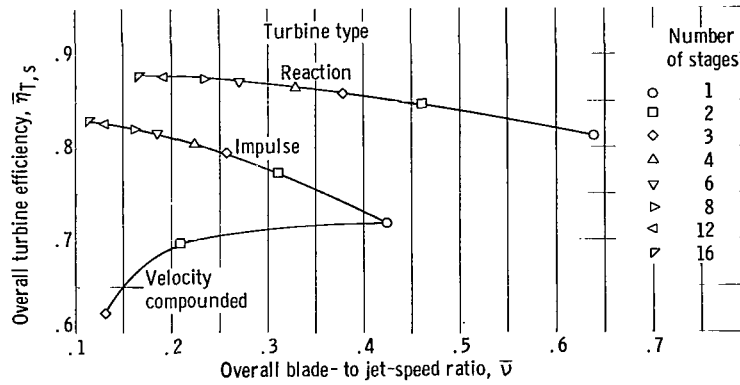


Figure 2. - Turbine aerodynamic efficiency characteristics. Constant of proportionality k (as defined in ref. 7), 0.4; Reynolds number, 10^6 ; mean-section stator-exit angle, 70° ; ratio of last-stage rotor- to stator-exit axial kinetic energies, 1.5.

The number of stages required for each turbine type and arrangement is related to blade speed through the characteristics presented in figure 2 and the definition of blade- to jet-speed ratio (eq. (2)). For each turbine considered, therefore, U/\bar{v} is obtained from equation (2) and multiplied by the appropriate value of \bar{v} obtained from figure 2 in order to find the mean-section blade speed U associated with any given number of stages. The mean-section blade diameter as a function of mean-section blade speed is obtained from

$$D = \frac{60 U}{\pi N} \quad (3)$$

and is plotted against blade speed in figure 3 for rotative speeds of 8000, 12 000, and 20 000 rpm.

The hub- to tip-radius ratio for the last stage is

$$\left(\frac{r_h}{r_t}\right)_n = \frac{1 - \frac{A_n}{\pi D^2}}{1 + \frac{A_n}{\pi D^2}} \quad (4)$$

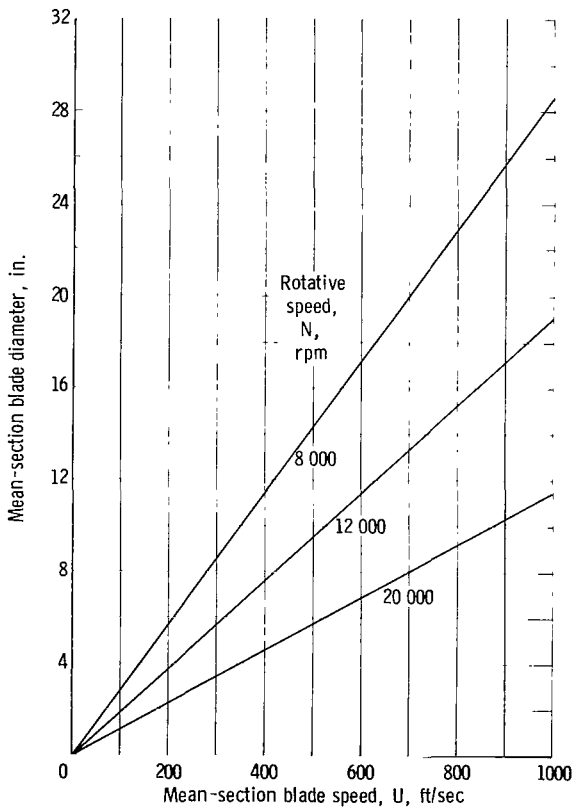


Figure 3. - Effect of blade speed on blade diameter.

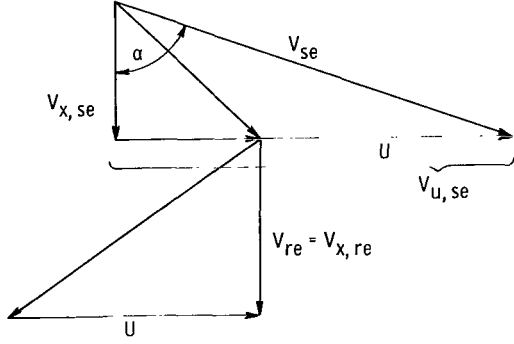


Figure 4. - Typical last-stage velocity diagram for stage speed-work parameter between 0.5 and 1.

where the exit annulus area A_n is

$$A_n = \frac{w}{\rho V_{x, re, n}} \quad (5)$$

Weight flow w is obtained from equation (1) and fluid density ρ from reference 5; both values were based on a turbine efficiency of 0.75 for the parametric analysis and on the moisture-corrected figure-2 values for the selected de-

signs. The axial velocity at the last-stage rotor exit is computed from the analysis assumptions and velocity diagram (fig. 4) geometry relations as follows:

$$V_{x, re, n} = \sqrt{E} V_{x, se, n}$$

$$V_{x, se, n} = V_{u, se, n} \cot \alpha$$

$$V_{u, se, n} = \Delta V_{u, n}$$

$$\Delta V_{u, n} = \frac{U}{\lambda_n}$$

Combining these equations yields

$$V_{x, re} = \sqrt{E} \cot \alpha \left(\frac{U}{\lambda_n} \right) \quad (6)$$

In accordance with the selected design criteria, $E = 1.5$, $\alpha = 70^\circ$, $\lambda_n = 1$ for reaction turbines, and $\lambda_n = 0.5$ for impulse and velocity-compounded turbines.

RESULTS OF ANALYSIS

Since there are no established blade-speed limitations for the prevention or control of erosion when alkali-metal working fluids are used, the turbine-geometry characteristics were determined parametrically as a function of blade speed for the various imposed design criteria. The parametric results are presented first in this section. After the results of the parametric analysis are discussed, the performance and geometry characteristics are presented in detail for several design points selected at low blade speeds.

TABLE I. - CALCULATED TURBINE PARAMETERS

[Inlet temperature, 2500° R; temperature ratio, 0.80; efficiency, 0.75.]

	Fluid											
	Sodium			Potassium			Rubidium			Cesium		
	Arrangement											
	Single	Dual		Single	Dual		Single	Dual		Single	Dual	
		Tur- bine 1	Tur- bine 2		Tur- bine 1	Tur- bine 2		Tur- bine 1	Tur- bine 2		Tur- bine 1	Tur- bine 2
Ideal specific work, $\overline{\Delta h}_{id}$, Btu/lb	338	171	182	168	84.3	89.5	67.1	34.8	36.3	40.5	20.6	21.6
Actual specific work, Δh , Btu/lb	254	128	137	126	63.2	67.1	50.3	26.1	27.2	30.4	15.5	16.2
Exit moisture, m	0.120	0.061	0.068	0.101	0.052	0.057	0.090	0.046	0.054	0.081	0.040	0.049
Weight flow, w, lb/sec	4.39	4.36	4.09	8.86	8.83	8.37	22.2	21.4	20.4	36.7	36.0	34.6
Exit density, ρ , lb/cu ft	0.0123	0.0351	0.0116	0.0645	0.1477	0.0615	0.211	0.440	0.203	0.365	0.729	0.352
Exit volume flow, w/ρ , cu ft/sec	357	124	353	137	59.8	136	105	48.6	101	101	49.4	98.2

Parametric Results

Some of the turbine parameters calculated for the parametric analysis are presented in table I. These values are presented in order to show the levels of specific work and flow encountered in these turbines and also to establish the basis for some of the observed geometry effects with respect to the fluid. The primary things to note from table I are the decrease in both specific work and volume flow with increasing fluid molecular weight (sodium, 23; potassium, 39; rubidium, 85; and cesium, 133). At a given blade speed, this decrease implies a smaller number of stages and a decreasing blade height (larger values of hub- to tip-radius ratio) with increasing molecular weight.

Number of stages. - The required number of turbine stages for the four fluids of interest are plotted against mean-section blade speed in figure 5 for both single- and dual-turbine arrangements using reaction, impulse, and velocity-compounded turbines. For the dual-turbine arrangement, the ordinate refers to the number of stages per machine, and thus the total number of stages for the arrangement is double that indicated. The curves of figure 5 reflect the effects of turbine arrangement, turbine type, fluid, and blade speed. The total number of stages required for the dual-turbine arrangement was slightly greater than that required for a single turbine. The number of stages is proportional to the specific work, which is slightly greater for the dual-turbine arrangement than for the single-turbine arrangement, as can be seen from table I. For the same blade speed, a reaction turbine requires about twice as many stages as an impulse turbine, whereas a velocity-compounded turbine requires half or less of that required for an impulse turbine. This effect is merely due to the basic characteristics of these selected turbine types as shown in figure 2 (p. 7), where an increase in number of stages is accompanied by an increase in turbine efficiency for the same blade- to jet-speed ratio.

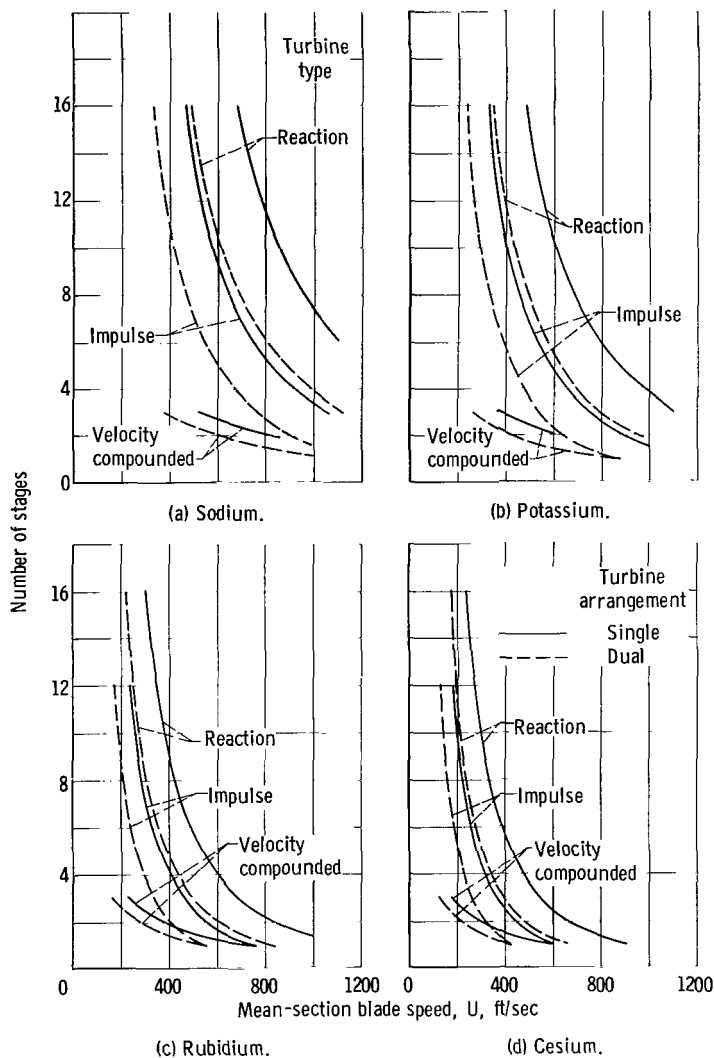


Figure 5. - Effect of blade speed on number of stages.

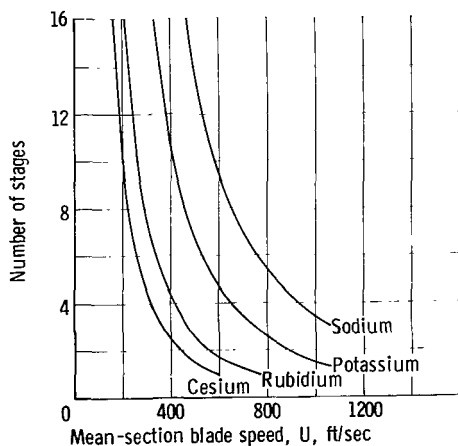


Figure 6. - Effect of fluid on number of stages. Turbine type, impulse; turbine arrangement, single.

The fluid effect discussed previously, a decrease in number of stages with an increase in molecular weight, is caused by the differences in fluid specific work. This effect is more clearly illustrated in figure 6, where the number of stages is plotted against the blade speed for a single impulse turbine using the four fluids. For reaction and impulse turbines, the required number of stages are in the ratio of about 9.0 to 4.5 to 1.7 to 1.0 for sodium, potassium, rubidium, and cesium, respectively (figs. 5 and 6). For velocity-compounded turbines, the fluid effect is much less pronounced; that is, the ratio of stages is less than 3 for sodium as compared with cesium. The effect of blade speed on number of stages is quite severe since stage work is proportional to the square of blade speed. For reaction and impulse turbines, the number of stages is inversely proportional to the square of blade speed, whereas for velocity-compounded turbines it is somewhat less than an inverse proportion. An impulse turbine in a single-turbine arrangement is used to illustrate the blade-speed effect (see fig. 6). For potassium, the required number of stages increases from about 2 to 10 as blade speed decreases from 900 to 400 feet per second; for cesium, there is an increase from about 1 to 10 stages as blade speed decreases from 600 to 200 feet per second.

Hub- to tip-radius ratio. - The

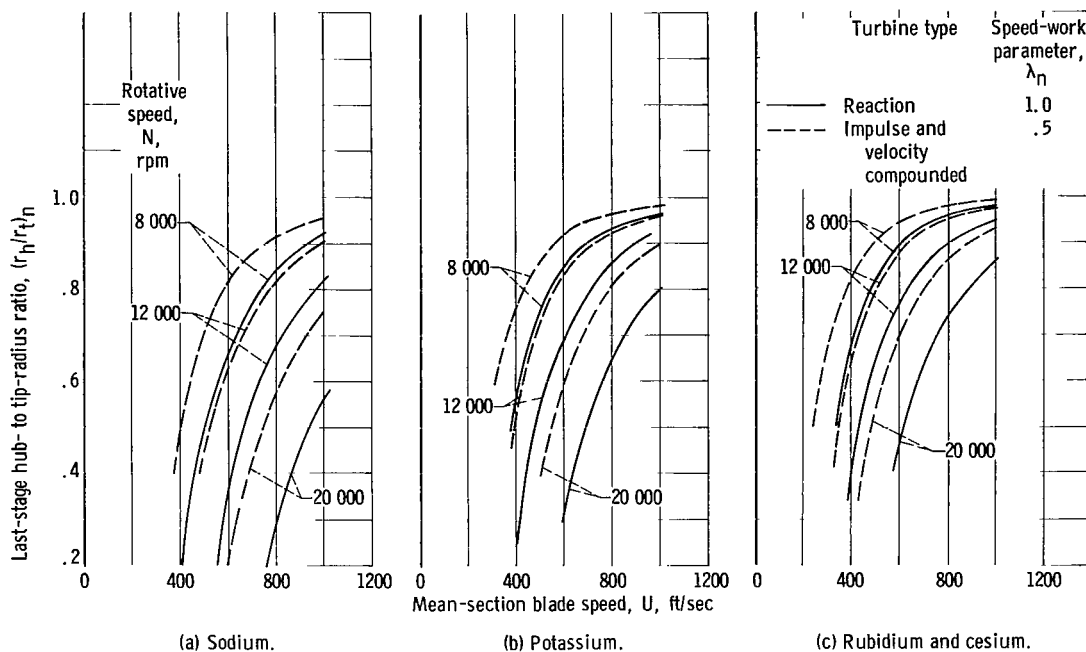


Figure 7. - Effect of blade speed on last-stage hub- to tip-radius ratio. Turbine efficiency, 0.75.

effect of mean-section blade speed on last-stage hub- to tip-radius ratio is presented in figure 7 for the four fluids of interest. These curves pertain to both the single turbine and the second machine of the dual-turbine arrangement. Rubidium and cesium are represented by the same set of curves because their volumetric flow rates differ by only a few percent (table I, p. 9). The curves in figure 7 are for rotative speeds of 8000, 12 000, and 20 000 rpm as well as for $\lambda_n = 1$ (reaction turbine) and $\lambda_n = 0.5$ (impulse and velocity-compounded turbines). The difference between the $\lambda = 1$ and the $\lambda = 0.5$ cases is due to the difference in associated exit-velocity levels (eq. (6)).

If a favorable geometry for the turbine is to be obtained, the last-stage hub- to tip-radius ratio should be larger than about 0.5 to 0.6. This consideration sets a lower limit on allowable blade speed (see fig. 7). This limit depends upon (1) the rotative speed, which for a given blade speed determines the mean-section blade diameter, (2) the speed-work parameter, which determines the nature of the proportionality between fluid velocity and blade speed, and (3) the working fluid, which determines the volumetric flow rate. In addition, the assumed values of α and E affect the hub- to tip-radius ratio, and usually these values may be adjusted to yield favorable blade geometries. For the lowest rotative speed (8000 rpm) considered herein and the highest fluid velocity (corresponding to $\lambda = 0.5$), the lower limit on mean-section blade speed, for the selected values of α and E , is about 400 feet per second for sodium, 300 feet per second for potassium, and 250 feet per second for rubidium and cesium. Increasing the rotative speed results in an increase in this minimum blade speed. With potassium, for example, an increase in rotative speed from 8000 to 12 000 rpm causes the minimum blade speed

to increase from about 300 to about 400 feet per second. The selection of a reaction rather than an impulse or a velocity-compounded turbine results in an increase of about 100 feet per second in the lower limit for blade speed.

For a given rotative speed and turbine type, increasing the blade speed results in a rapid increase in hub- to tip-radius ratio. With potassium, for example, increasing the blade speed from 400 to 600 feet per second for an impulse turbine rotating at 12 000 rpm results in an increase in the last-stage hub- to tip-radius ratio from 0.54 to 0.83. In order to obtain a desirable hub- to tip-radius ratio, the rotative speed must depend upon the specified blade speed and vice versa (fig. 7).

Selected Turbine Designs

In order to illustrate the geometry and performance characteristics of turbines designed to operate with low blade speeds, several typical design points were selected as a result of the parametric analysis. For these illustrative selections, only two of the four previously considered working fluids were used. Potassium was selected because there is much current interest in potassium systems, and cesium was selected because it is the fluid that yields the minimum number of stages. The geometry and performance characteristics for the design selections are presented in detail in tables II(a) and (b) for potassium and cesium, respectively. These selections were made primarily on the basis of low blade speed, with the rotative speed so chosen as to yield a reasonable geometry. Mean-section blade speeds in the range of about 300 to 500 feet per second were selected. The selection of this blade-speed range can be considered arbitrary; the only reasoning was that the erosion problem is reduced as blade speed is reduced. Six typical selections were made for each of the two fluids.

For potassium (table II(a)), the selections included a single three-stage and dual two-stage 8000-rpm velocity-compounded turbines, a single ten-stage and dual six-stage 8000-rpm impulse turbines, dual four-stage 12 000-rpm impulse turbines, and dual nine-stage 8000-rpm reaction turbines. Turbine efficiency varied from about 55 percent for the single velocity-compounded turbine to about 80 percent for the dual reaction turbines. The mean-section blade diameters ranged from about 9 to 13 inches. For cesium (table II(b)), the selections included a single two-stage 8000-rpm velocity-compounded turbine, a single two-stage and dual one-stage 12 000-rpm impulse turbines, dual two-stage 8000-rpm impulse turbines, and a single five-stage and dual three-stage 8000-rpm reaction turbines. Turbine efficiency varied from about 64 percent for the single velocity-compounded turbine to about 80 percent for the dual reaction turbines. The mean-section blade diameters ranged from about 8 to 12 inches. These characteristics indicate that reasonable geometry and performance can be obtained even if the blade speed is limited to values as low as 300 to 500 feet per second.

TABLE II. - CHARACTERISTICS OF TYPICAL TURBINE DESIGNS

(a) Fluid, potassium

	Turbine type									
	Reaction		Impulse						Velocity compounded	
			Arrangement							
	Dual		Single	Dual				Single	Dual	
	Turbine 1	Turbine 2		Configuration 1		Configuration 2			Turbine 1	Turbine 2
Turbine 1				Turbine 2	Turbine 1	Turbine 2				
Inlet temperature, °R	2500	2250	2500	2500	2250	2500	2250	2500	2500	2250
Inlet pressure, lb/sq in. abs	165	75	165	165	75	165	75	165	165	75
Exit temperature, °R	2250	2000	2000	2250	2000	2250	2000	2000	2250	2000
Exit pressure, lb/sq in. abs	75	29	29	75	29	75	29	29	75	29
Number of stages	9	9	10	6	6	4	4	3	2	2
Rotative speed, rpm	8000	8000	8000	8000	8000	12 000	12 000	8000	8000	8000
Ideal specific work, Btu/lb	84.3	89.5	167.8	84.3	89.5	84.3	89.5	167.8	84.3	89.5
Actual specific work, Btu/lb	68.8	72.6	122.2	64.2	67.8	62.5	66.1	93.6	55.1	58.2
Exit moisture	0.059	0.064	0.096	0.054	0.058	0.052	0.055	0.063	0.043	0.047
Weight flow, lb/sec	8.11	7.63	9.11	8.69	8.22	8.93	8.47	11.91	10.12	9.69
Exit density, lb/cu ft	0.1488	0.0620	0.0642	0.1480	0.0616	0.1477	0.0614	0.0619	0.1462	0.0609
Exit volume flow, cu ft/sec	54.5	123.1	141.9	58.7	133.4	60.5	137.9	192.5	69.2	159.1
Blade- to jet-speed ratio	0.220	0.220	0.144	0.183	0.183	0.222	0.222	0.132	0.210	0.210
Efficiency	0.816	0.811	0.728	0.762	0.758	0.741	0.738	0.558	0.654	0.650
Mean-section blade speed, ft/sec	459	459	419	382	382	464	464	384	438	438
Mean-section blade diameter, in.	13.2	13.2	12.0	11.0	11.0	8.8	8.8	11.0	12.6	12.6
Reynolds number	0.97×10 ⁶	0.96×10 ⁶	1.20×10 ⁶	1.25×10 ⁶	1.24×10 ⁶	1.60×10 ⁶	1.59×10 ⁶	1.71×10 ⁶	1.27×10 ⁶	1.28×10 ⁶
Last-stage hub- to tip-radius ratio	0.866	0.724	0.783	0.877	0.739	0.843	0.675	0.649	0.900	0.789

(b) Fluid, cesium

	Turbine type								
	Reaction			Impulse					Velocity compounded
				Arrangement					
	Single	Dual		Single	Dual				Single
		Turbine 1	Turbine 2		Configuration 1		Configuration 2		
				Turbine 1	Turbine 2	Turbine 1	Turbine 2		
Inlet temperature, °R	2500	2500	2250	2500	2500	2250	2500	2250	2500
Inlet pressure, lb/sq in. abs	216	216	115	216	216	115	216	115	216
Exit temperature, °R	2000	2250	2000	2000	2250	2000	2250	2000	2000
Exit pressure, lb/sq in. abs	50	115	50	50	115	50	115	50	50
Number of stages	5	3	3	2	1	1	2	2	2
Rotative speed, rpm	8000	8000	8000	12 000	12 000	12 000	8000	8000	8000
Ideal specific work, Btu/lb	40.5	20.6	21.6	40.5	20.6	21.6	20.6	21.6	40.5
Actual specific work, Btu/lb	31.7	16.7	17.4	28.4	14.0	14.6	15.1	15.7	25.9
Exit moisture	0.087	0.047	0.055	0.071	0.034	0.040	0.039	0.046	0.058
Weight flow, lb/sec	35.2	33.4	31.8	39.3	39.7	38.4	37.0	35.6	43.1
Exit density, lb/cu ft	0.367	0.735	0.354	0.361	0.725	0.349	0.728	0.351	0.355
Exit volume flow, cu ft/sec	95.9	45.4	89.8	108.9	54.8	110.0	50.8	101.3	121.4
Blade- to jet-speed ratio	0.295	0.378	0.378	0.304	0.420	0.420	0.304	0.304	0.210
Efficiency	0.783	0.813	0.805	0.702	0.683	0.677	0.734	0.727	0.639
Mean-section blade speed, ft/sec	420	389	389	432	432	432	313	313	299
Mean-section blade diameter, in.	12.0	11.1	11.1	8.2	8.2	8.2	8.9	8.9	8.5
Reynolds number	3.26×10 ⁶	3.34×10 ⁶	3.34×10 ⁶	5.32×10 ⁶	5.38×10 ⁶	5.46×10 ⁶	4.62×10 ⁶	4.65×10 ⁶	5.64×10 ⁶
Last-stage hub- to tip-radius ratio	0.720	0.823	0.679	0.681	0.825	0.678	0.812	0.658	0.558

CONCLUDING REMARKS

This study shows that geometrically feasible turbines can be obtained even with blade speeds limited to values below 500 feet per second. Designs featuring a very low number of stages (as low as one) can be achieved. These designs, however, are associated with a poorer, but possibly acceptable, performance than are those designs featuring a large number of stages (possibly greater than 10). Considerations such as the fluid, the turbine type and arrangement, and the blade speed all affect the nature and level of the relation between the number of stages and the efficiency.

The selection of a particular design becomes the choice of the system designer, who must work within the constraints of his system. These constraints may include such ones as a specified fluid, a required performance, or a packaging limitation. The minimum-stage designs can be overhung, with one hot bearing thus eliminated, and easily packaged. If maximum performance is a major design criterion, the larger number of stages must be tolerated and the mechanical problems associated with longer shafts must be considered.

SUMMARY OF RESULTS

This study was undertaken to determine the general geometry characteristics of alkali-metal turbines as they are affected by blade-speed limitations that might result from moisture-caused erosion effects. Cycle selection was made on the basis of reduced moisture levels in the turbine. Dual-turbine systems with intermediate moisture separation as well as single turbine systems were studied. Rotative speeds were so selected as to be compatible with common alternator frequencies. Turbine characteristics, such as number of stages, diameter, and exit hub- to tip-radius ratio, were determined for 1-megawatt power systems using sodium, potassium, rubidium, and cesium as working fluids at a turbine-inlet temperature of 2500° R. The pertinent results of this study are summarized as follows:

1. The selection of a turbine type depends upon the balance between the number of stages and the performance desired for any particular system. For a given blade speed, the reaction turbine required about twice as many stages as did the impulse turbine. The velocity turbine required about half or less the number of stages required for the impulse turbine. The offsetting factor is the increase in achievable efficiency associated with the turbine types requiring an increased number of stages.

2. For reaction and impulse turbines, the required number of stages was in the ratio of about 9.0 to 4.5 to 1.7 to 1.0 for sodium, potassium, rubidium, and cesium, respectively. For velocity-compounded turbines, the fluid effect was much less pronounced; the ratio of stages was less than 3 for sodium as compared with cesium.

3. For reaction and impulse turbines, the number of stages was inversely proportional to the square of the blade speed, whereas for velocity-compounded turbines it was somewhat less than an inverse proportion to the blade speed. For a single impulse turbine with potassium, for example, the required number of stages increased from about 2 to 10 as the blade speed decreased from 900 to 400 feet per second.

4. The requirements for a favorable turbine diameter and hub- to tip-radius ratio set a lower limit on blade speed for any given rotative speed. For the assumed blade configuration parameters and the lowest rotative speed (8000 rpm) considered herein, this lower limit on mean-section blade speed was about 400 feet per second for sodium, 300 feet per second for potassium, and 250 feet per second for rubidium and cesium. This blade-speed limit increased with increasing rotative speed.

5. Geometrically feasible turbines were obtained even with blade speeds limited to values below about 500 feet per second. Designs featuring a very low number of stages (as low as one) can be evolved. These designs, however, are associated with a lower efficiency, which may be about 60 percent or less, than are those designs featuring a large number of stages (possibly greater than 10), which may achieve efficiencies of about 80 percent.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 19, 1965.

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